University of Technology, Sydney

Faculty of Engineering and IT

STUDY OF DRAG TORQUE IN A TWO-SPEED DUAL CLUTCH TRANSMISSION ELECTRIC VEHICLE POWERTRAIN SYSTEM

A thesis submitted for the degree of

Doctor of Philosophy

Xingxing Zhou

(September 2014)

CERTIFICATE OF ORIGINAL AUTHORSHIP

I certify that the work in this thesis has not previously been submitted for a degree nor has it been submitted as part of requirements for a degree except as fully acknowledged within the text.

I also certify that the thesis has been written by me. Any help that I have received in my research work and the preparation of the thesis itself has been acknowledged. In addition, I certify that all information sources and literature used are indicated in the thesis.

Signature of Student:

Zhou Xingxing

XINGXING ZHOU

Date:

ACKNOWLEDGEMENTS

My thesis could not have come to fruition without the assistance of many contributors. I would like to thank my principal supervisor Professor Nong Zhang. He supported me with great direction, advices and any other helps in my research. His knowledge, hard-working and research attitude fostered my development in various aspects. I would like to thank my co-supervisor Dr. Paul. David Walker. He provided me with great guidance during my whole candidature period. He is the most brilliant and helpful co-supervisor to me. I would also press my gratitude to A/Prof Jinchen Ji and Dr. Zhen Luo who often offered their time discussing research issues in my thesis. I would like thank Chris Chapman and Holger Roser for their technical support in building test rig and doing experiments.

I would like to thank my colleagues, Bo Zhu, Jiageng Ruan, Yu Wang, Fei Ding, Sangzhi Zhu, Tianxiao Zhang, Yiqiang Wang, Guangzhogn Xu, Nga Hoang, Jin Zhao, Lifu Wang, Robert W. Heal, Li Sun, Erbo Chen, Di Cao for their support.

I also wish to gratefully acknowledge the consistent financial support of the following agents: China Scholarship Council (CSC), and University of Technology, Sydney (UTS).

Finally, I would like to thank all my family for standing with me through all these years as well.

TABLE OF CONTENTS

LIST OF FIGURES

LIST OF TABLES

NOMENCLATURE

Global abbreviations used in this thesis

DCT = dual clutch transmission
MT = manual transmission
AMT = automated manual transmission
CVT = continuously variable transmission
EV = electric vehicle
HEV = hybrid electric vehicle
EM = electric machine

Chapter 3 Notation

TCU = transmission control unit
VCU = vehicle control unit
DCT = dual clutch transmission
m_V = vehicle mass (kg)
α = vehicle acceleration (m/s^2)
r_t = tyre radius (m)
T_{EM} = electric machine torque (N.m)
γ = gear ratio
T_V = vehicle torque (N.m)
C_R = coefficient of rolling resistance
g = gravity (m/s^2)
φ = road incline (Rad)
C_D = drag coefficient
ρ = air density (kg/m^3)
A_V = vehicle frontal area(m^2)
V_V = vehicle speed (m/s)
P_R = vehicle resisting power (W)
ω_V = vehicle rotational speed (rad/s)

Chapter 5 Notation

A_g = arrangement constant for gearing
b_w = face width in contact (mm)
C1 = clutch 1
C2 = clutch 2
D = outside diameter of the gear (mm)
d_m = diameter (mm)
E_{DCT} = efficiency of DCT
F = total face width (mm)

F_b = applied load (N)
f = turbulent flow coefficients
f_0 = bearing dip factor
f_g = gear dip factor
f_m = mesh coefficient of friction
f_L = bearing load empirical factor
Hs = sliding ratio at the start of approach action
Ht = sliding ratio at the end of recess action
h = tangential line velocity modifying exponent
h_c = clutch plate's clearance (mm)
h_{con} = length of concentric shaft (mm)
j = viscosity modifying exponent
K = load intensity (N/mm^2)
L = length of the gear (mm)
M = mesh mechanical advantage
M_t = transverse tooth module
N = number of frictional surface
n_1 = pinion rotational speed (rpm)
n_{motor} = motor speed (rpm)
P_B = power losses caused by bearings drag torque (KW)
P_{Con} = power losses caused by concentric shaft drag torque (KW)
P_{bl} = load independent power loss (KW)
P_{bv} = speed independent power loss (KW)
P_{Ch} = power losses caused by gear churning (KW)
P_{Cl} = power losses caused by wet clutch plates drag torque (KW)
P_G = power losses caused by gear meshing drag torque (KW)
P_{GW} = individual gear windage and churning loss (KW)
P_L = total power losses in DCT (KW)
P_{oil} = oil seal power losses (KW)
Q = flow rate (m3/s)
T_1 = pinion torque (N.m)
T_B = drag torque caused by bearings (N.m)
$T_{B(1,2)}$ = drag torque caused by bearing (1) and (2) (N.m)
T_{Cl} = drag torque caused by wet clutch packs (N.m)
T_{Ch} = drag torque caused by churning (N.m)
T_{con} = drag torque caused by concentric shafts viscous shear resistance (N.m)
T_{1st_output} = output torque of the outer concentric shaft (N.m)
T_{1st_output} = output torque of the inner concentric shaft (N.m)
T_{final_outpu} = final output torque from DCT (N.m)
T_{GM} = drag torque caused by gear pairs meshing (N.m)
$T_{GM_{1st\_pair}}$ = drag torque caused by 1^{st} gear pair meshing (N.m)
T_m = motor output torque (N.m)
V = pitch line velocity (m/s)
V_{oil} = velocity for oil seal (rpm)
R_{con_i} = outer radius of the inner shaft (mm)
R_{con_o} = inner radius of the outer shaft (mm)
R_f = roughness factor

R_{o2} = gear outside radius (mm)
R_{w2} = gear operating pitch radius (mm)
R_{o1} = pinion outside radius (mm)
R_{w1} = pinion operating pitch radius (mm)
r = gear ratio
r_m = mean radius (mm)
r_o = outer radius of the clutch (mm)
r_{1st} = 1^{st} gear ratio
r_{2nd} = 2^{nd} gear ratio
z_1 = number of pinion teeth
z_2 = number of gear teeth
α_w = transverse operating pressure angle (^{o})
β = generated helix angle (^{o})
β_w = operating helix angle (^{o})
μ = viscosity of the oil ((Ns/m^2))
ν = kinematic oil viscosity (m^2/s)
ρ_{oil} = density of oil (kg/m^3)
ω_{EM} = output speed of the electric motor (rpm)
$\Delta\omega$ = relative speed between two concentric shafts (rpm)
$\Delta\omega_{c1}$ = relative speed within clutch 1 (rpm)
$\Delta\omega_{c2}$ = relative speed within clutch 2 (rpm)
∇p = pressure difference between the input and output of clutch pair (pa)

Chapter 7 Notation

A_{ca} = surface area of transmission case (m^2)
A_{oil} = oil-side surface area of transmission case (m^2)
c_{oil} = specific heat of oil (J/kg.K)
c_{DCT} = the average specific heat of transmission (J/kg.K)
h_{ca} = height of gearbox housing (m)
G_r = grashoff number
k = heat transmission coefficient (W/m^2K)
m_{oil} = mass of oil
P_B = power losses caused by bearings drag torque (kW)
P_{Con} = power losses caused by concentric shaft drag torque (kW)
P_{Ch} = power losses caused by gear churning (kW)
P_{Cl} = power losses caused by wet clutch plates drag torque (kW)
P_G = power losses caused by gear meshing drag torque (kW)
P_L = total power losses in DCT (kW)
Q_{Ca} = quantity of heat dissipated by the transmission case (J)
Q_{all} = overall quantity of DCT heat (J)
Q_{oil} = quantity of heat absorbed by oil (J)
Q_{DCT} = quantity of heat absorbed by DCT structure (J)
R_e = Reynold's number
T_{oil} = oil temperature (Kelvin)

T_{en} = environment temperature (Kelvin)
T_0 = the primary temperature (Kelvin)
T_{wall} = temperature of housing wall (Kelvin)
α_{oil} = oil-side heat transfer coefficient (W/m^2K)
α_{ca} = air-side heat conduction (W/m^2K)
α_{con} = convection part(W/m^2K)
α_{rad} = radiation part (W/m^2K)
α_{free} = heat transfer coefficient due to free convection (W/m^2K)
α_{forced} = heat transfer coefficient due to forced convection (W/m^2K)
λ_{wall} = coefficient of thermal conduction of housing (W/mK)
δ_{wall} = mean housing wall thickness (m)
ε = emission ratio
η^* = temperature ratio
ν_{air} = impingement velocity (m/s)
ΔT = increased temperature caused by absorbing heat (Kelvin)

Chapter 8 Notation

A_V = vehicle frontal area(m^2)
C_D = drag coefficient
C_R = coefficient of rolling resistance
E_{DCT} = efficiency of DCT
E_{motor} = efficiency of motor
f_{obj} = objective function
g = gravity ($m.s^2$)
m_V = vehicle mass (kg)
N = number of surfaces of clutch plates
P_R = vehicle resisting power (W)
P_B = power losses caused by bearings drag torque (KW)
P_{Con} = power losses caused by concentric shaft drag torque (KW)
P_{Ch} = power losses caused by gear churning (KW)
P_{Cl} = power losses caused by wet clutch plates drag torque (KW)
P_G = power losses caused by gear meshing drag torque (KW)
P_L = total power losses in DCT (KW)
p_{max} = clutch plates maximum permissible pressure (MPa)
r_t = tyre radius (m)
r_i = inner radius of clutch plates (m)
r_o = outer radius of clutch plates (m)
T_{EM} = electric machine torque (N.m)
T_V = vehicle torque (N.m)
T_{max} = maximum torque transmitted by clutch (N.m)
V_V = vehicle speed (m/s)
V_{shift} = final shift speed point (km/h)
V_{index} = index of original shift point (km/h)
γ = gear ratio
γ_{first} = the 1^{st} gear ratio

γ_{second} = the 2nd gear ratio
α = vehicle acceleration (m/s^2)
μ = dynamic viscosity (N/m^2)
φ = road incline (Rad)
ω_v = vehicle rotational speed (rad/s)
ρ = air density (kg/m^3)

ABSTRACT

Pure electric vehicles are widely looked as a potential avenue to reduce fossil fuel consumption and emission in the long term in the transportation section. Pure electric vehicles currently being used in the market are mainly equipped with single speed transmissions, with tradeoffs between dynamic (such as climbing ability, top speed, and acceleration) and economic performance (drive range). The employment of two-speed dual clutch transmission (DCT) in electric vehicles is likely to improve average motor efficiency and range, and even can reduce the required motor size. In order to comprehensively improve the electric vehicle powertrain system efficiency, it is necessary to fully consider the proposed two-speed transmission drag torque, including its influences and potential applications.

The focus of this thesis is on studying the drag torque within two-speed dual clutch transmissions. Different source of drag torque in the DCT are theoretically analysed and modelled, including torsional resistance caused by viscous shear caused between wet clutch plates and concentrically aligned shafts, gear mesh friction and windage, oil churning, and bearing losses. Then experimental works are carried out on UTS electric vehicle powertrain system test rig. Outcomes of experimentation on drag torque confirm that simulation results agree well with the test data. Then, based on the drag torque study, the thermal behaviour of the transmission is analysed via both theoretical and experimental investigation. Finally, integrated optimal design of electric vehicle powertrain system equipped with two-speed DCT is performed, considering drag torque, shift schedule, electric motor selection, gear ratio design and wet clutch design.